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Air Source Heat Pump for Northern Climates Part I: Simulation of Different Heat Pump Cycles

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AIR SOURCE HEAT PUMP FOR NORTHERN CLIMATES PART I: SIMULATION OF DIFFERENT HEAT PUMP CYCLES

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ABSTRACT

Air source heat pumps are widely used for residential heating because of the relatively low installation costs. Major disadvantages are that the heat output and COP decrease and the discharge temperature of the compressor increases as the outdoor temperature decreases. All of these factors usually lead to the need of combined heat pump and backup heating systems, which increases the cost and lowers the efficiency of the overall system.

In this study, a novel air-source, two-stage heat pump using R-410A as the refrigerant was simulated, designed, constructed and tested for ambient temperatures as low as -30°C and supply temperatures of up to 50°C. In addition to air and water heating, the system is also able to provide sufficient air conditioning in cooling mode.

The study presented here is divided into two parts. The first part entitled "Simulation of Different Heat Pump Cycles" summarizes the results of an extended literature and patent review, and presents a theoretical analysis of the three most promising cycles. The second part is entitled "Measurement and Verification" and focuses on the design, implementation, and testing of a breadboard system as well as on the comparisons with commercially available heat pumps.

1. INTRODUCTION

Heating requirements in northern climates, such as in Minnesota, South and North Dakota, and Manitoba, can represent a challenge for air source heat pumps, since the ambient air temperature in this region can reach values as low as -30°C during the heating season. Nevertheless, most heat pumps for space heating in this area are still air source heat pumps due to their ease of installation and their low installation costs. The four main problems at very low ambient temperatures for air source heat pumps are:

a) Insufficient heat output as the required heat is the largest whereas the heat pump capacity is reduced due to lower mass flow rates of the refrigerant
b) High compressor discharge temperature caused by the low suction pressure and high pressure ratio across the compressor. This leads to bivalent heat pump systems which use the heat pump at medium ambient temperatures. At very low ambient temperatures the compressor is turned off to protect it from overheating and electric resistance heating is used instead. The combined efficiency of such systems is quite low.
c) The coefficient of performance (COP) decreases rapidly for high pressure ratios which can be found at low ambient temperature conditions.
d) If the heat pump is designed for low ambient temperature conditions it will have a capacity that is far too large at medium ambient temperatures. Therefore, the heat pump needs to cycle on and off at higher ambient temperatures in order to reduce its capacity, which leads to a lower efficiency of the system compared to steady-state performance and also a lower comfort level of the inhabitants.

Several concepts have been proposed to address these problems and to design heat pumps for low temperature climates. A comparison of many of these possibilities can be found in *Bertsch et al. (2005),* which summarizes and compares the advantages and disadvantages of eight concepts. These concepts include two-stage compression systems with either intercooling, economizing or in a cascade arrangement, systems which inject two-phase refrigerant into the compressor to decrease the compressor discharge temperature (*Zogg 2002)* and systems which use high oil flow rates to cool the compressor.

Table 1 *(Bertsch et al. 2005)* lists the data of seven concepts for a comparison. The single stage cycle was used as a reference cycle with 100% relative heat output and 100% relative efficiency, even though the single stage cycle does not work for the given application since the discharge temperatures of the compressor would be too high. Two stage cycles can be built with either one two-stage compressor or two single-stage compressors. The first concepts shows less lubrication issues since the compression mechanism is covered in one shell, whereas the second concept with two different compressors allows more freedom to design the system and also to only run one compressor at low load conditions. Both the intercooler cycle and economizer cycle are able to achieve a relatively high efficiency combined with the ability to vary the load by only using single stage compression. The discharge temperature stays within the limits over the whole operating range. A cascade cycle shows probably the best performance at low temperatures, since the usage of different refrigerants for the two cycles allows more optimization potential. Negative aspects of the cascade system are difficult adaptations to higher ambient temperatures where the compressors will unload due to a low pressure ratio and to air conditioning mode. Refrigerant injection has some potential to extend the operating range, without changing the efficiency drastically. However, it will not be able to cover the operating range for temperatures as low as -30°C ambient temperature. Also, oil cooling can be used to lower the discharge temperature. The main issue with oil cooling is the reduction of the volumetric efficiency of the compressor and the increased superheat due to mixing of warm oil with the refrigerant at the compressor inlet. Since the suction volume at such low temperatures is already very high, it will be difficult to find appropriate compressors. Finally, mechanical subcooling is a good way to increase the performance of the system, especially in cases when two different sink temperatures are needed, but the main problem of overheating the compressors can not be solved.

Table 1: Comparison of different heat pump cycles

#	Concept	Preferred Compressor*)	Number heat output steps	Relative Efficiency	Relative Heat output	Discharge temperature
1	1-stage cycle	LT	1	100%	100%	High
2	2-stage w. intercooler	2-stage	1	130%	100%	Acceptable
		Sc, Recip, Rot	3	130%	140%	Acceptable
3	2-stage w. economizer	2-stage	1	130%	100%	Low
		Sc, Recip, Rot	3	130%	150%	Low
4	Cascade cycle	Sc, Recip, Rot	1	140%	140%	Low
5	Refrigerant injection	Sc, Screw	2	Comparable	115%	High
6	Oil cooling	Recip, Rot	1	Comparable	Comparable	Acceptable
7	Mechanical subcooling	LT + Sc	2	110%	120%	High

*) Sc…Scroll, Recip…Reciprocating, Rot…Rotary, LT…Low temperature

Based on the comparison of the seven concepts listed in Table 1, it was concluded that the cascade cycle, the two-stage cycle with inter-cooling and the two-stage cycle with economizing are the most feasible concepts for the given application. Again, the operating conditions of the system in heating mode range from ambient temperatures as low as -30°C to 10°C with heat supply temperatures of 40°C to 50°C. In air conditioning mode the indoor temperature is relatively constant at 25°C whereas the ambient temperature ranges from 25°C to 40°C.

2. DESCRIPTION OF THE 3 MOST PROMISING CONCEPTS

2.1 Two-stage heat pump with intercooler

The two-stage heat pump with intercooling consists of two compressors that are arranged in series with a heat exchanger in between the discharge line of the first one and suction line of the second one. The schematic of this cycle is shown in Figure 1 together with the corresponding pressure-enthalpy diagram in Figure 2. As explained above, the cycle matches the given operating conditions well and can be used with two single-stage compressors as shown in the schematic or with one two-stage compressor. The intercooler decreases the refrigerant inlet temperature of the high pressure stage compressor which decreases also the discharge temperature of the same compressor. This measure extends the operation of the heat pump over a wider outdoor temperature range. Studies by several authors *(Zehnder and Favrat 2000*, *Zogg 1999)* showed that the oil migration in such systems is a critical issue, especially if only one compressor is running. Therefore, the usage of an oil separator or other measures to ensure even oil distribution between both compressors is recommended.

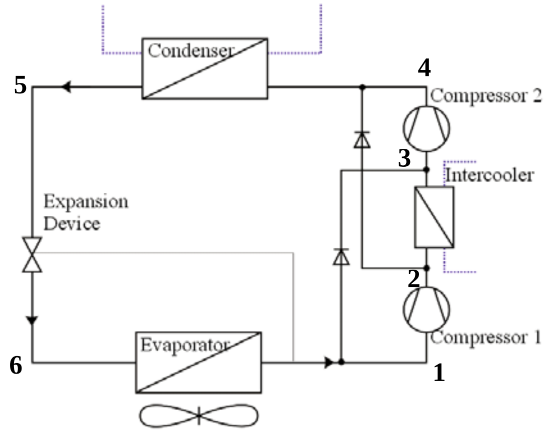

Figure 1: Schematic of a 2-stage cycle with intercooler and without suction line heat exchanger

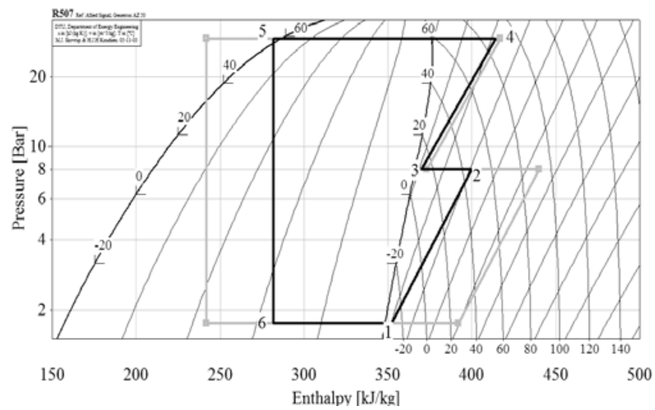

Figure 2: P-h-diagram for the 2-stage cycle with intercooler (Black line with suction line heat exchanger and grey line without)

2.2 Two-stage heat pump with economizer

Instead of using an intercooler, a two-stage cycle may be operated with a closed economizer as shown in a simplified schematic in Figure 3. Figure 4 shows the cycle state points of a two-stage heat pump with economizer in a pressure-enthalpy diagram. The economizer delivers a certain amount of two-phase refrigerant to a mixing chamber in the suction line of the high pressure-stage compressor where it is mixed with the hot discharge gas of the low pressure-stage compressor. This measure allows very accurate control of the inlet state of the high pressure-stage compressor. Therefore, the compressor discharge temperature of the high pressure-stage compressor should be low. In addition, the economized refrigerant subcools the remaining condensed refrigerant before it enters the expansion valve, which improves the system COP. Instead of using a closed economizer, an open economizer could be used *(Zogg 1999).*

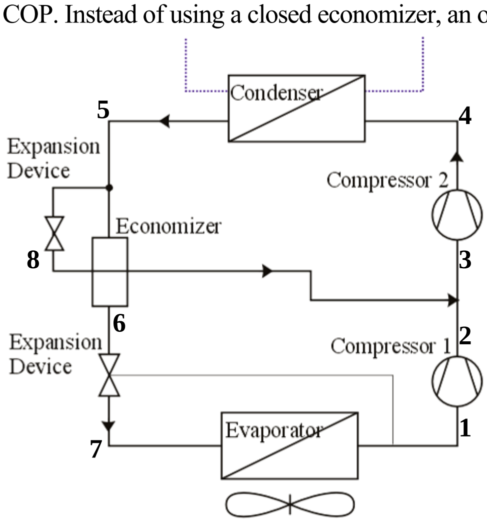

Figure 3: Schematic of a two stage cycle with Economizer

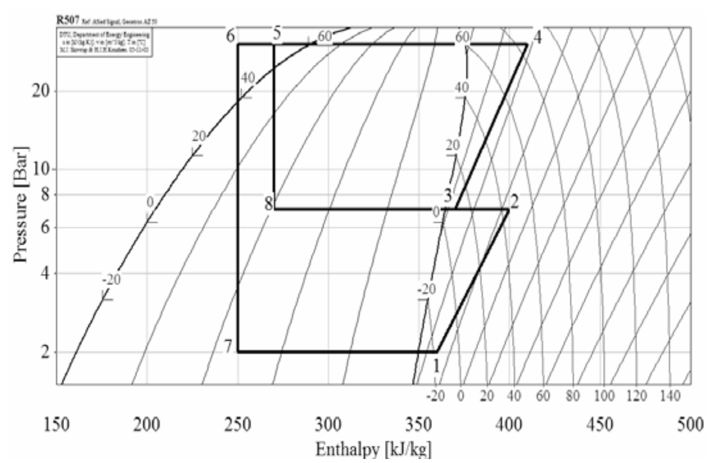

Figure 4: p-h-diagram for the 2-stage cycle with economizer

2.3 Cascade cycle

Cascade cycles have been used in many different refrigeration applications for a long time. Figure 5 shows a schematic of the system and Figure 6 the corresponding pressure-enthalpy diagram. In the case presented in Figure 6, both stages use the same refrigerant, which greatly reduces the potential of this system. The main advantage of the cascade cycle over other two-stage cycles is that two different refrigerants can be used whereby every refrigerant works at its optimum design range.

Most low temperature refrigerants have disadvantageous properties at high temperatures and vice versa. Whereas the system COP at high pressure ratios is very good, it decreases for higher ambient temperatures due to the temperature gap in the intermediate heat exchanger and the lower pressure ratio across the compressors. Another disadvantage is that it is more difficult to reverse the cycle or run in part load since the system is not able to run one compressor alone. A possible solution to this issue would be to add an additional outdoor-evaporator to the high temperature stage cycle, which could be used in single-stage mode. The cascade system shows altogether very good performance and leaves a lot of room for further improvement. Good efficiency at high temperature lifts and no oil management problems may be able to compensate for higher installation and maintenance costs. For the given simulations an additional outdoor heat exchanger was assumed so the high stage cycle (5, 6, 7, 8) can be operate without the low stage.

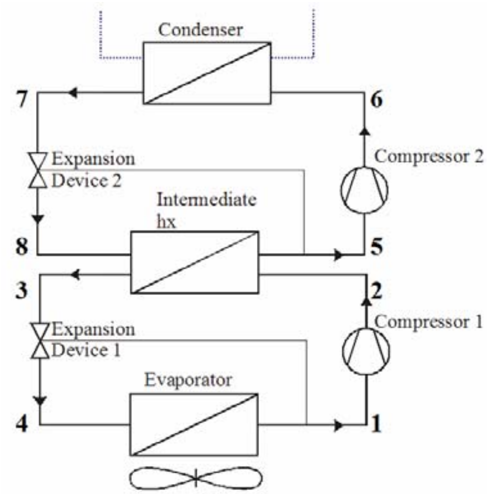

Figure 5: Schematic of a cascade cycle

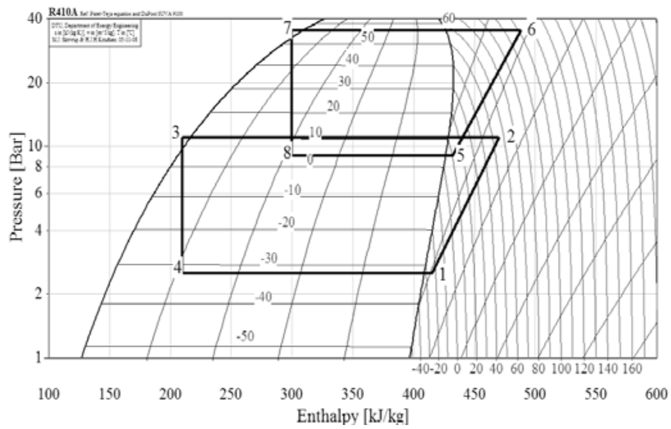

Figure 6: P-h-diagram of a cascade cycle

3. DETAILED COMPARISON OF THE THREE MOST PROMISING CYCLES

3.1 General Approach

The simulation was approached using a steady state model consisting of part models for each component of the cycle. These component models could then be arranged and used for all three heat pump cycles. In this way the results should be comparable with only little error. A general assumption for all simulations was the usage of dry air for the ambient air. This means that frost buildup and therefore, defrosting, was not considered in the simulations. Furthermore, all pressure drops and heat losses in the refrigerant lines were neglected. It is assumed that these simplifications will affect all three cycles in a similar way and therefore, they have little influence on the results of the comparison.

3.2 Compressor model

Since no data for commercial compressors is available for the operating range of this application, a simplified physical model (*Mackensen, 2002*) was used to extend performance data of some available standard compressors. In the usual operating range, the standard 3rd order polynomial approach according to ANSI/ARI Standard 540-1999 was used due to its higher accuracy. In order to account for variations of the superheat the following approach by Dabiri and Rice (*1981*) was used to correct the refrigerant mass flow rate:

$$\dot{m}_R = \dot{m}_{R\_map} \cdot \left[1 + F_1 \cdot \left(\frac{\rho_{new}}{\rho_{map}} - 1\right)\right] \tag{1}$$

with $\rho_{new} = f(P_1,T_1)$, $\rho_{map} = f(P_1,T_{1\_map})$, and F_1=0.75 (recommended by author)

With the updated refrigerant mass-flow-rate, the electrical input power can also be corrected to account for the difference in superheat between compressor map and simulation.

$$P_{el} = P_{el\_map} \cdot \frac{\dot{m}_R}{\dot{m}_{R\_map}} \cdot \left[\frac{h_{2s} - h_1}{h_{2s\_map} - h_{1\_map}}\right] \tag{2}$$

The heat loss of the compressor is calculated using a fixed heat loss factor and the compressor exit state is determined by using an energy balance across the compressor.

In order to account for the needed compressor size, a size factor was introduced which implies that compressors can be built with similar performance over a certain size range. By changing the compressor size a constant heat rate of 17 kW at the condenser could be achieved at the design point of -10°C ambient temperature and 50°C supply temperature.

3.3 Heat exchanger model

A heat pump with water cooled condenser was considered for the simulations. Assuming a fixed amount of subcooling, the heat transfer rate of the condenser can be equated as follows:

$$\dot{Q}_{cond} = \dot{m}_R \cdot \left(h_3 - h_2\right) \tag{3}$$

where h_3 is evaluated at the condensing pressure and a temperature calculated with a the fixed subcooling.

Then, the effectiveness, ε_c, of the condenser is calculated knowing the heat transfer characteristics of the heat exchanger and the water specific heat:

$$\varepsilon_c = 1 - e^{-NTU_c} \quad \text{with} \tag{4}$$

$$NTU_C = \frac{U \cdot A_c}{c_w \cdot \dot{m}_w} \tag{5}$$

Using the condenser effectiveness, the updated condensing temperature, T_{cond}, and the water outlet temperature, $T_{w,out}$, can be calculated as follows:

$$T_{cond} = T_{w,in} + \frac{\dot{Q}_{cond}}{\varepsilon_c \cdot \dot{m}_w \cdot c_w} \tag{6}$$

$$T_{w,out} = T_{w,in} + \frac{\dot{Q}_{cond}}{\dot{m}_w \cdot c_w} \tag{7}$$

A very similar approach was used to calculate the evaporating temperature and the air outlet temperature at the outdoor unit (evaporator) as well as for the suction line heat exchanger and internal heat exchangers.

3.4 Expansion device model

The expansion device was assumed adiabatic which leads to an isenthalpic process:
$h_4 = h_3$

3.5 Closing of the cycles

All component models were combined within two iteration schemes. For example, the cascade system first calculates one cycle and then the other using the inputs form the first one. It achieves the final results by iterating the two loops. For the intercooler and the economizer model, the part system with the two compressors and the intercooler / economizer is iterated first and then the results of the whole cycle are used to update the first iteration. At the beginning, the suction and discharge condition are guessed and then the final result is iterated by updating the pressures throughout several runs of the overall iteration loop. Convergence of the systems is normally quite fast and stable.

4. RESULTS

Simulations were carried out for several ambient and indoor temperatures and combined in diagrams, which are presented in this section. For example, Figure 7 shows the simulation results of the Coefficient of Performance (COP) versus the outdoor temperature for the economizer cycle. The solid line shows the results for two-stage heating, which in the given case has the best performance for outdoor temperatures below -5°C compared to single stage heating. The difference between the efficiency of heating with the high pressure-stage compressor only compared to the low pressure stage compressor only is mainly due to the difference in heat transfer rate at the condenser and evaporator, which favors the high pressure-stage heating.

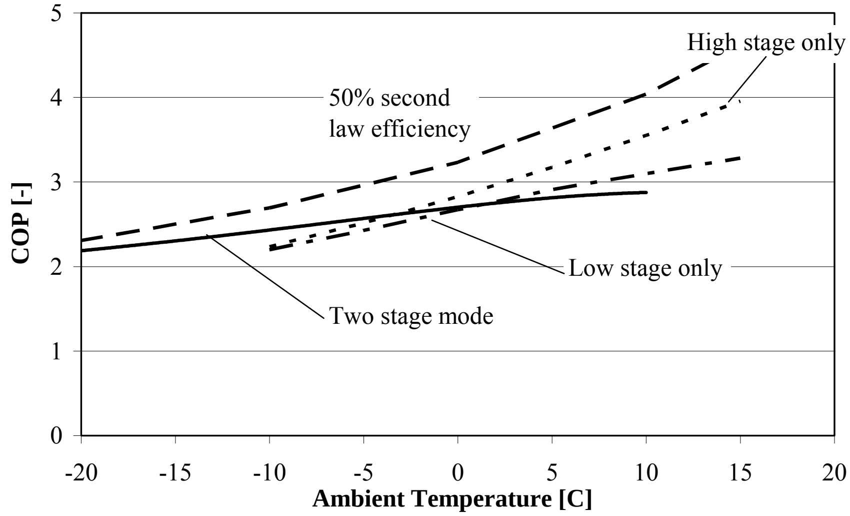

Figure 7: Simulation results for COP of the economizer cycle at 50°C supply temperature versus ambient temperature

As can be seen, the COP of the economizer cycle is corresponds to a second law efficiency of approximately 40% in single-stage mode and close to 50% in two-stage mode, which is very high compared to standard heat pump systems.

The condenser heat rejection rate of the economizer cycle is presented as a function of the outdoor temperature in Figure 8. As expected the capacity in two stage mode is the highest and the capacity in single-stage mode using the high pressure stage compressor is lowest. The high pressure-stage compressor has approximately 50% of the theoretical suction volume of the low pressure-stage compressor, which explains that the capacity running the low pressure-stage compressor alone is about twice the capacity of the high pressure-stage compressor.

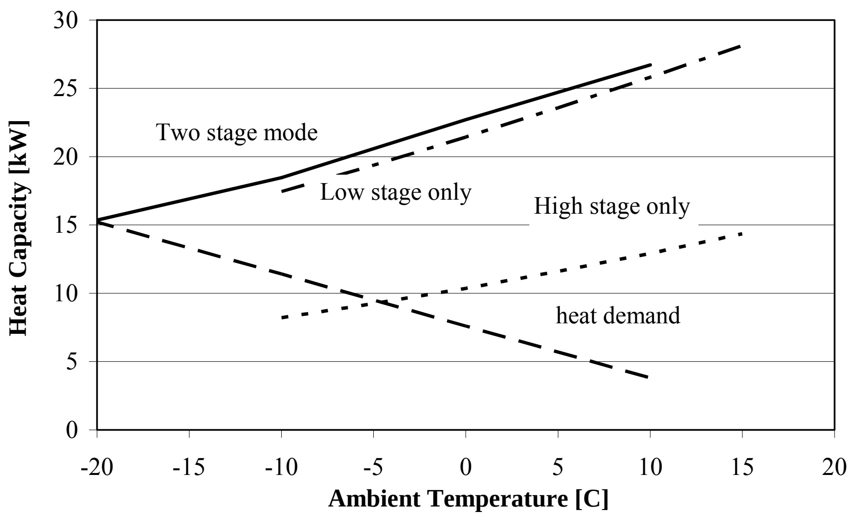

Figure 8: Simulation results for heating capacity of the economizer cycle at 50°C supply temperature versus ambient temperature

In a real system, the operating mode of single-stage or two-stage is set according to the heating demand. The switch-over between the operating modes needs to consider minimal and maximal compression ratios for both compressors and the discharge temperature of the high pressure-stage compressor.

The presentation of the simulation results of the cascade cycle and intercooler cycle are omitted in favor of the comparison of the three cycles. Figure 9 shows the COP of the cascade, intercooler and economizer cycles at a supply temperature of 50°C versus the ambient temperature. For simplicity the switchover point from single-stage to two-stage modes is assumed at 0°C outdoor temperature, which explains the step of the COP in the diagram. The dotted line indicates a COP that corresponds to 50% of the second law efficiency as a reference. The cascade cycle shows the best performance towards lower ambient temperatures whereas the intercooler and economizer cycle have a better performance in single-stage mode. The main problem of the intercooler cycle for the given application is that the return temperature of the water is too high for a successful cooling of the hot discharge gas of the low pressure-stage compressor. Especially at low ambient temperatures, the 2nd law efficiencies of the cascade and economizer cycle are higher than the ones of the intercooler cycle. Even in single-stage mode, the efficiencies of the cascade and economizer cycle are high in comparison to commercially available heat pumps mainly due to the large heat exchangers that are used as the evaporator and condenser.

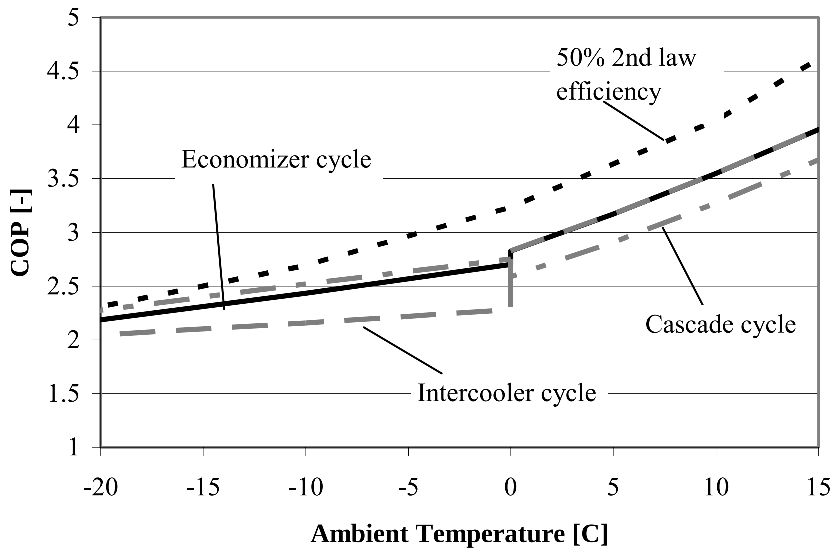

Figure 9: Comparison of the COP for a supply temperature of 50°C versus ambient temperature of the three cycles

Figure 10 shows the heating capacity of the 3 cycles at a supply temperature of 50°C versus the ambient temperature. As can be seen from Figure 10, the curves are very similar to each other and fit the heating demand quite well. The single-stage mode using the low pressure-stage compressor only is not shown since its capacity is too large for the application range it can be used in. In addition, its COP is lower than the one of single-stage heating with the high pressure-stage compressor or two-stage heating at low ambient temperatures.

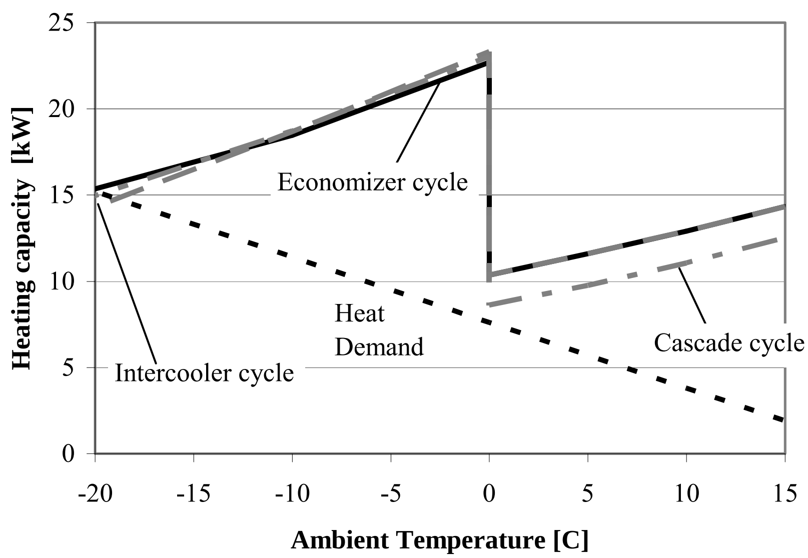

Figure 10: Comparison of the heating capacity for a supply temperature of 50°C versus ambient temperature of the three cycles

5. CONCLUSIONS

There are several possibilities to design residential air source heat pumps for low temperature climates. Some of these possibilities are described and compared in this paper. All of the cycles presented in this paper have their advantages and disadvantages and their usage may depend on the exact specifications of the given installation. For the three most promising cycles, the economizer cycle, the intercooler cycle and the cascade cycle in depth simulations were carried out to decide which system best fits the operating conditions. The results of these simulation favor the economizer cycle due to its good performance and easy controllability compared to the other two cycles. Even though the cascade cycle shows marginally better performance at low ambient temperatures the cycle is more complex to build and therefore, more expensive. Since the cost difference between intercooler and economizer cycle is small, and the performance of the intercooler cycle is lower than the one of the economizer cycle, the economizer cycle seems to be the best choice considering performance parameters and costs.

The second part of this paper shows the design, measurement and validation of the simulation model as well as a comparison to commercially available heat pumps.

6. NOMENCLATURE

Symbol	Description	Unit	Index	Description
COP	Coeff. of Performance	[-]	1	State-point compressor inlet
ε	Efficiency	[-]	2	State-point compressor outlet
h	Enthalpy	[kJ/kg]	3	State-point condenser outlet
$\dot{m}$	Mass-flow-rate	[kg/s]	4	State-point evaporator inlet
P	Electrical power	[kW]	c, cond	Condenser
ρ	Density	[kg/m^3]	map	According to compressor map
T	Temperature	[°C]	R	Refrigerant
Q	Heat rate	[kW]	w	Water

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